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**Technical Report No. 32-196**

**Recirculation of a Two-Phase Fluid  
by Thermal and Capillary Pumping**

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PASADENA, CALIFORNIA**

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## ABSTRACT

A closed-cycle gas-supply system for gas bearings and gas-floated devices is described which eliminates mechanical pumps or compressors and uses instead thermal and capillary pumping action. A small quantity of a two-phase fluid of suitable thermodynamic characteristics, such as Freon, is recirculated in a closed system. The fluid is thermally vaporized in an evaporator, and the superheated vapor, after passing through the gas bearing, is condensed and returned to the evaporator by capillary action. The system is of special interest to space applications, because it can operate in a zero-*g* environment from solar or nuclear power sources, without conversion to electrical energy.

## I. THE CLOSED-CYCLE SYSTEM

The lubrication and flotation of rotating, sliding, or pivoting masses by compressed gases or vapors offer significant advantages. These have long been recognized and utilized, particularly in applications involving machinery and instruments operating under unusual conditions of environment and speed or stringent requirements of precision. Typical examples are bearings exposed to extremes of temperatures, both low and high, or to

nuclear radiation, which may cause conventional lubricants or flotation liquids to break down. The interest in gas lubrication has been greatly stimulated recently by the need for more accurate and reliable inertial instruments, particularly gyros for missiles and spacecraft applications as well as rotating or sliding computer devices (for instance, memory drums and magnetic readout components).

In some of these applications, self-acting (hydrodynamic) bearings which do not require a separate source for the supply of gas can be used advantageously. However, the load-carrying capacity of a self-acting bearing is limited, as is its life, when frequent starts and stops are encountered. These limitations do not apply to externally pressurized (hydrostatic) gas bearings, which have found many useful applications where a supply of compressed air or other gases is conveniently available. Their use in missiles and boosters, however, had to be confined to missions of short duration in which the supply of compressed gas could readily be stored aboard in suitable containers. Storage of an expendable gas becomes impractical in spacecraft for lunar or planetary exploration because of the prohibitive size and weight of the container required for missions of long duration.

In applications of this type, recirculation of the gas becomes mandatory. It can be accomplished by mechanical pumps or compressors, driven by an electric motor which is supplied from the secondary power source on board. However, it can be shown that such a system imposes a substantial penalty in weight and power as a result of the losses incurred in converting solar or nuclear energy first into electrical and subsequently into mechanical energy.

For instance, an efficiency of 30% for motor-driven pumps or compressors (of the size involved) and of 10% for the conversion of solar into electric energy results in an over-all efficiency of only 3%. It is thus evident that a system which utilizes thermal energy directly for the regenerative transport of a gas or vapor, without conversion to electrical and mechanical energy, should produce significant savings in power and weight.

These considerations have led to the development of a closed-cycle supply system employing thermal "pumping" of a superheated vapor and capillary action for the return of the condensate to the evaporator. A regenerative system of this type is shown schematically in Fig. 1. A fluid of suitable thermodynamic properties is vaporized by input of heat in an evaporator and raised to the required operating pressure, which is controlled by a pressure regulator. To insure that no liquid enters the bearing, the vapor is superheated by an auxiliary heater, as shown, or by lowering its pressure in an orifice before it enters the gas bearing. After passing through the bearing, the vapor is condensed, by lowering its temperature, and the condensate is "pumped" back to the evaporator by capillary action of a wick.

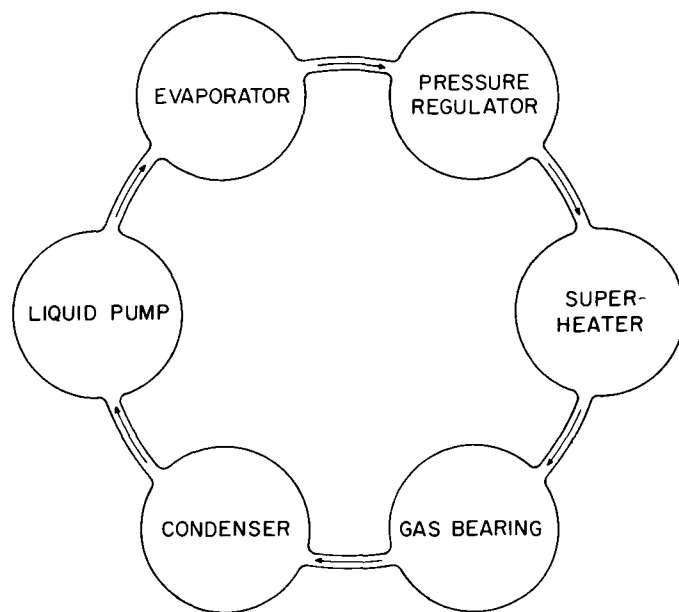


Fig. 1. Closed-cycle system

In the selection of a suitable fluid for a closed-cycle system of this kind, its thermodynamic properties must be considered, along with its effect on the materials of which the gas bearing is constructed, including gaskets and other materials exposed to it. The fluid should be nonexplosive, nonflammable, of low toxicity, and chemically very stable. To minimize the energy required to raise it to operating pressure, its specific heat and heat of vaporization should be low, and it should have low viscosity in the vapor phase to minimize the friction torque and losses in the bearing, which are proportional to viscosity.

From the fluids which meet these requirements, a number of Freon compounds were selected and used in this investigation. A Freon system can be optimized readily within the temperature ranges encountered. Figure 2 shows the pressure-temperature relationship of various Freon compounds manufactured by E. I. DuPont de Nemours and Company. To suit different conditions of application and environment, the temperature range can be shifted. The operating pressure of a gas bearing depends on the required load-carrying capacity and stiffness. For instance, if a pressure of 50 psig is required, Freon-113 may be used and its temperature raised to 214°F, or Freon-114 may be selected at a temperature of 122°F, as will be noted in Fig. 2.

The flow rate which a closed-cycle system has to deliver depends on the number and type of gas bearings

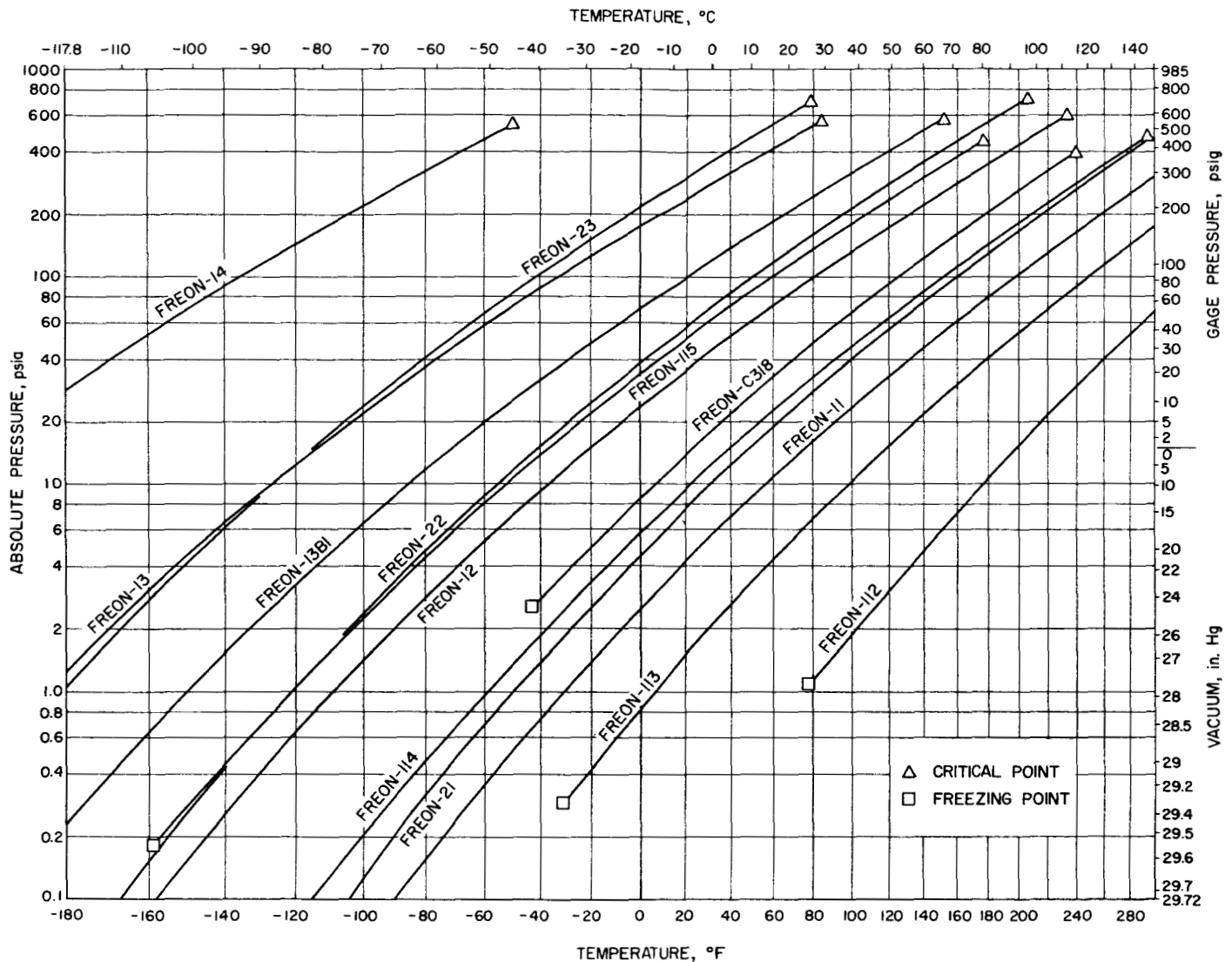


Fig. 2. Pressure-temperature relationships of Freon compounds

(or other consumers) connected to it. A typical journal bearing for the support of the gimbal axis of a single-axis gyro may consume from approximately 1/10 to 1 cfm at pressures between 1 and 40 psig, depending on the number of orifices and its *g*-capability. A gas-floated spinning sphere, used for the attitude control of a spacecraft or as a stable reference of an inertial system, can be operated with a substantially lower flow rate (i.e., approximately 0.05 cfm or less), since it can be supported by a small number of single-orifice pads.

The thermal power input of a closed-cycle system of this type for a given mass flow rate can be calculated fairly accurately by breaking it down into the following steps:

1. The power required to raise the liquid from the sump temperature in the condenser to the boiling temperature in the evaporator
2. The power of enthalpy increase between the saturated liquid and the saturated vapor at operating pressure
3. The power input for superheating
4. The total heat power loss of the system to its surroundings
5. The power required to return the condensate to the evaporator

It can be shown that the total power is a linear function of the mass flow rate  $W$ . The boiling temperature of a given Freon compound for a desired operating pressure can be read from the chart (Fig. 2). The increase in enthalpy at this temperature for the transition from the liquid to the saturated vapor phase can be obtained from pressure-enthalpy diagrams which the DuPont Company has published for various Freon products and of which Fig. 3 is an example for Freon-114. In order to arrive at an estimate of the heat loss (step 4), the loss was assumed to be proportional to the difference between the temperature of the vapor and ambient temperature. The power under step 5 is negligible.

In Fig. 4, the computed values of the total power required (steps 1 to 5) are plotted for three flow rates; i.e., 0.0478, 0.0342, and 0.0211 cfm (solid lines). It can be seen that the correlation with measured data (squares, circles, and triangles) is quite satisfactory. These data were taken on an experimental closed-cycle system, which

is shown in Fig. 5. A dome-shaped vessel is half filled with liquid Freon, into which a pancake-shaped Chromalox heater coil is immersed. The power input to the heater could be regulated and measured by a wattmeter (see Fig. 6). Variable-orifice-type flowmeters measured the flow rates of both the liquid and the vapor phase. The temperature and pressure at various locations in the system were recorded by means of thermocouples and pressure transducers. The gas bearing was simulated in this test rig by a regulating valve, whose rate of flow and pressure drop could be adjusted within the required rate. It was, together with the evaporator and gas flowmeter, enclosed in a wooden housing filled with vermiculite for better insulation. The heat loss of the system was determined experimentally and found indeed to be proportional to the difference between the vapor and room temperatures, as was assumed in the previous calculation.

Referring to Fig. 4, it can be seen, for example, that a thermal power of 20 w or less is sufficient to operate a

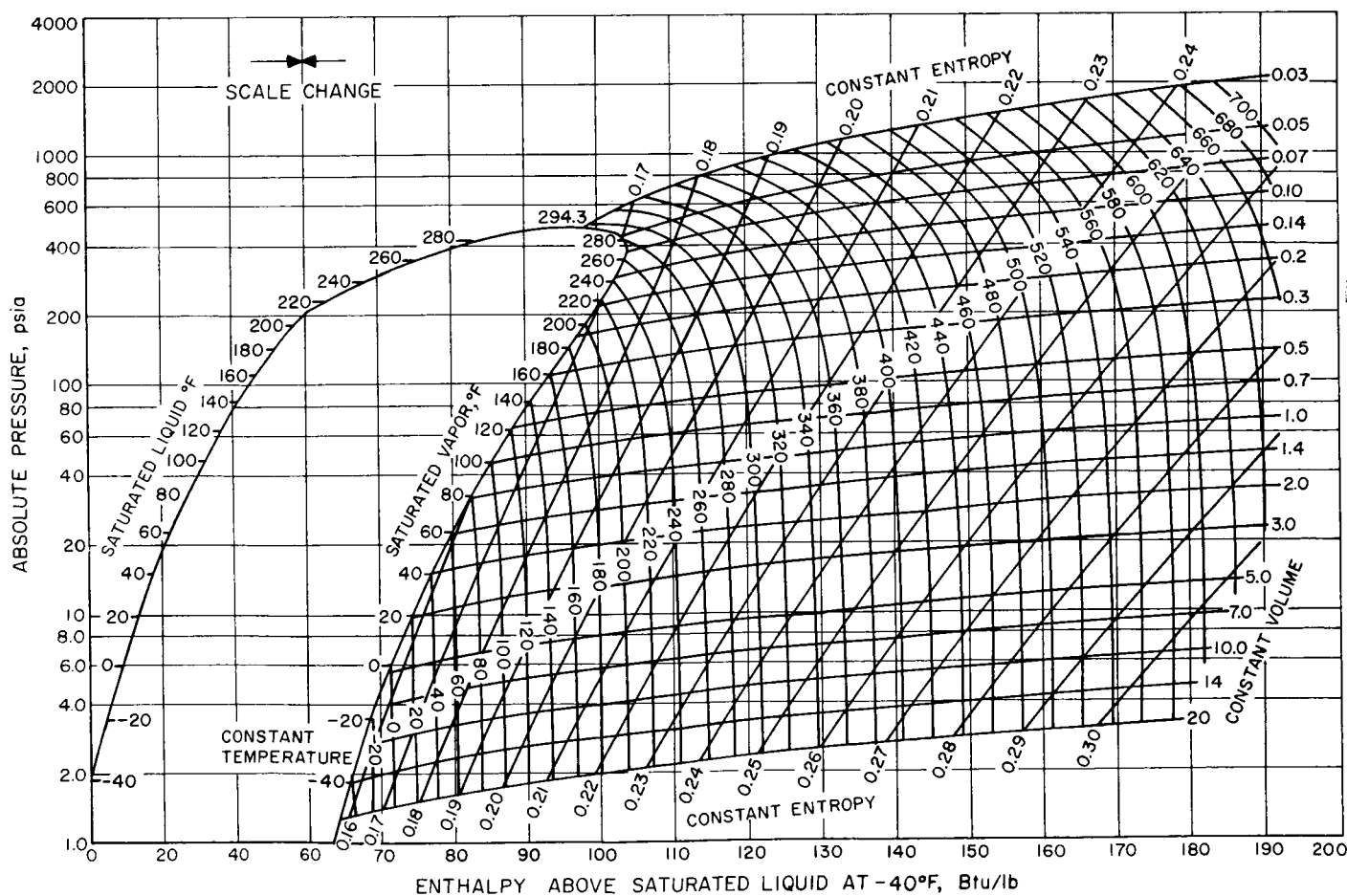
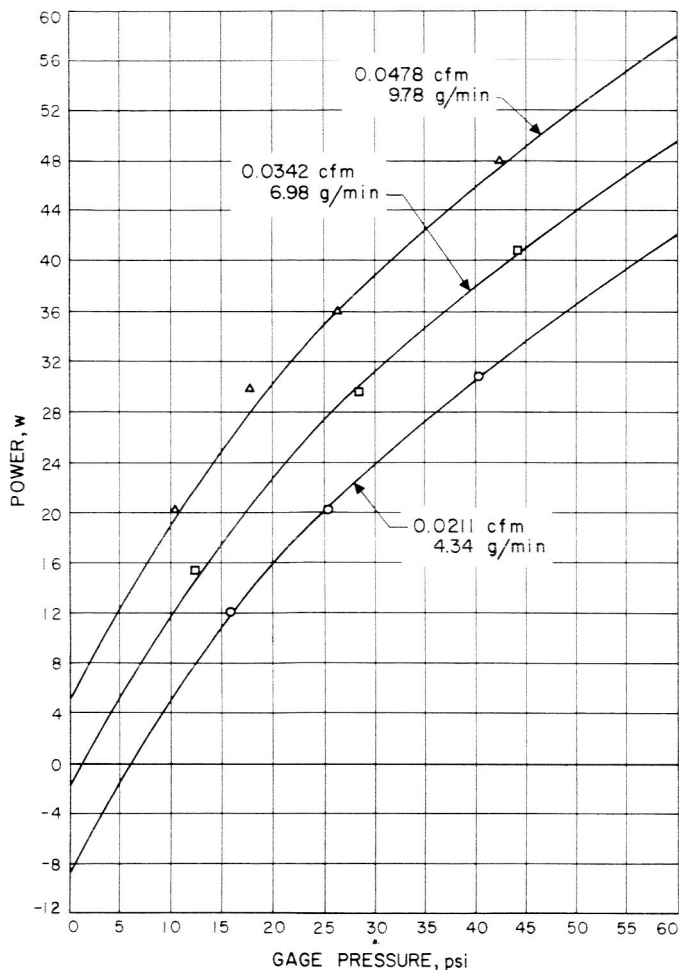


Fig. 3. Freon-114 pressure-enthalpy curves

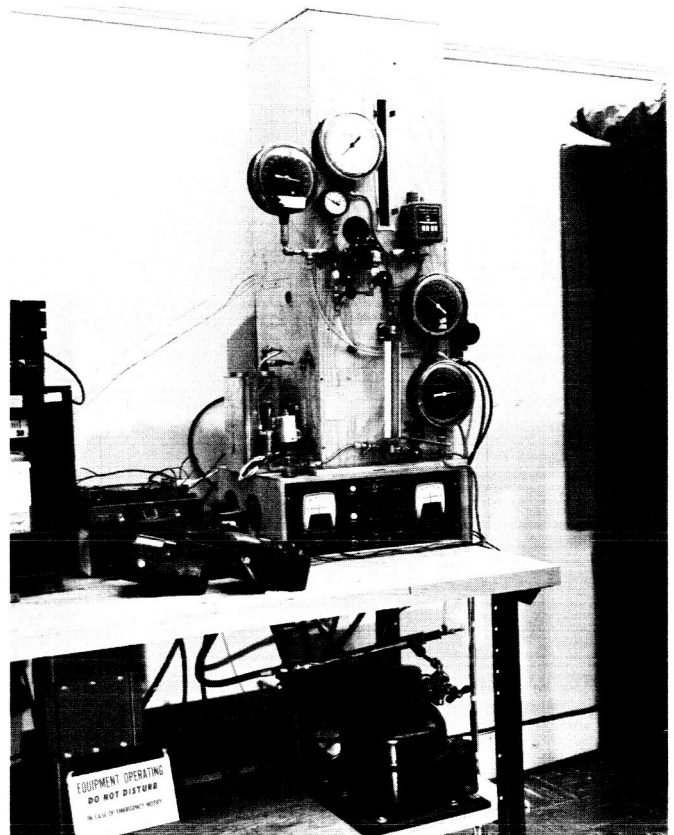




**Fig. 4. Freon-114 thermal-pumping-power requirements; comparison of predicted and measured values**

gas-floated spinning sphere supported by six single-orifice pads at a supply pressure of 10 psig. Since this thermal power can be supplied readily without conversion by solar panels or concentrators in a spacecraft, a closed-cycle system using Freon or a fluid with similar thermodynamic properties appears to be feasible for such applications as far as power requirements are concerned. The fluid is condensed by exposing it to the dark (cold) side of the space vehicle, and the condensate is returned to the evaporator by a mechanical pump or by capillary action, as will be explained later. It can be shown that the pumping power for the small flow rates involved is considerably less than 1 w; i.e., it is negligible.

The analysis and experiments so far described proved that a recirculatory system of this type is thermodynamically sound. However, it appeared desirable to investigate the performance of an actual gas bearing in a Freon system, instead of simulating it by a regulating valve.



**Fig. 5. Closed-cycle gas-supply experimental arrangement**

A second test apparatus was therefore constructed which incorporated a gas bearing of the single-orifice circular-pad type. A schematic diagram of this test set is shown in Fig. 7 and a photograph in Fig. 8. A stainless-steel evaporator tank was partially filled with liquid Freon. An electric heating element inside the tank was used to raise the liquid to its boiling point and vaporize it at the required rate. The power input was measured by a wattmeter. The vapor leaving the evaporator was saturated; heat losses between the evaporator and the bearing would cause it to be of reduced quality. To insure that no liquid would enter the control orifice of the bearing, a superheater, consisting of a small coil of resistance wire, was placed just upstream of the bearing orifice, over which the Freon vapor passed. Its power input was measured by another wattmeter. Temperatures were measured at four positions, as indicated in Fig. 7.

The bearing and loading fixtures were similar to the arrangement described in a paper by one of the authors (Ref. 1), except that the lower bearing surface was formed by one face of an optical prism. This permitted visual observation of the bearing surface; if there was any liquid

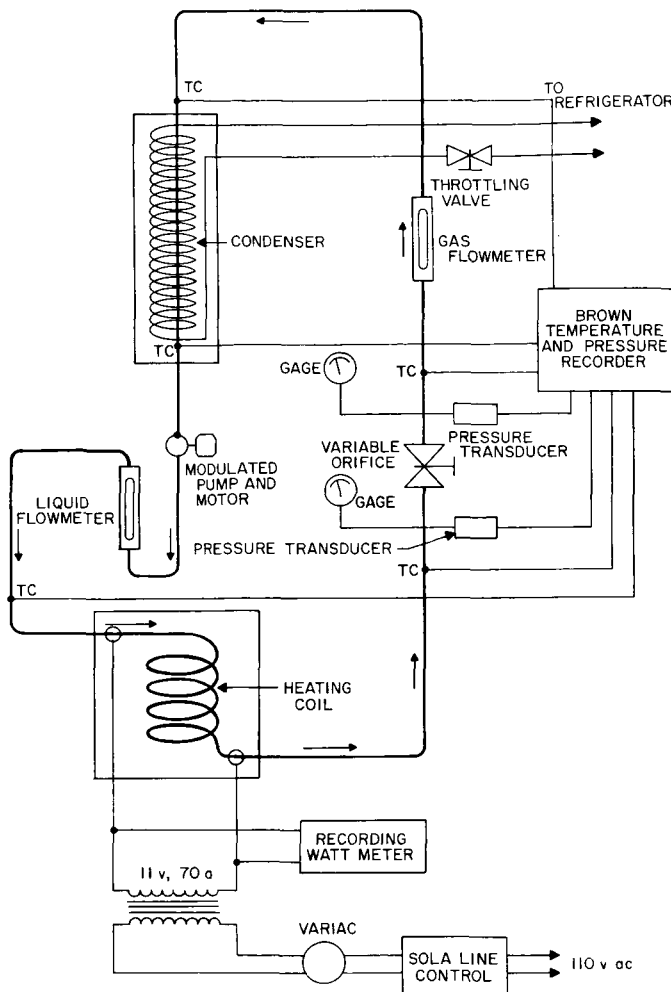


Fig. 6. Two-phase closed-cycle system

flow through the bearing, it could be detected quite readily. The prism also served as a very useful interferometer. By directing monochromatic light into the vertical face of the prism, a fringe pattern could be seen (see Fig. 9). The spacing of adjacent fringes is a measure of parallelism of the two bearing surfaces. By adjusting leveling screws at the base of the fixture, a bearing alignment not more than  $11 \mu\text{in.}$  out of parallel could be obtained easily. This degree of alignment is most important, because the minimum nominal bearing gap to be tested was  $175 \mu\text{in.}$  Reference capacitors, described previously, were used to determine the gap height.

The thrust of the bearing was opposed by a pneumatic load piston, which was prevented from cocking and made virtually frictionless by being supported radially in a journal gas bearing. The load was measured by a strain-gage-type dynamometer, placed between the bearing and the load piston. The flow rate of the Freon was deter-

mined by a rotameter, calibrated in terms of mass flow, between the bearing exit and the condenser.

Before operating the bearing on Freon, several runs were first made using nitrogen gas to verify the predicted performance. The experimental results agreed very well with the theory. Subsequent experiments with Freon vapor could therefore be expected to be valid and to show up any deviation from the analytical treatment of Freon vapor as an ideal gas.

One of the objectives of the tests was to determine whether Freon vapor in a slightly superheated state could operate a gas bearing without causing condensation in the bearing. In the preliminary experiments, it was found that the superheater, located just upstream of the orifice (see Fig. 7), could not reliably prevent some liquid flow through the orifice when the evaporator temperature was substantially higher than the ambient temperature. Apparently, the heat capacity of the bearing was too large for the small superheater to overcome. Therefore, another technique was developed which proved to be satisfactory and should be more readily adaptable to spacecraft appli-

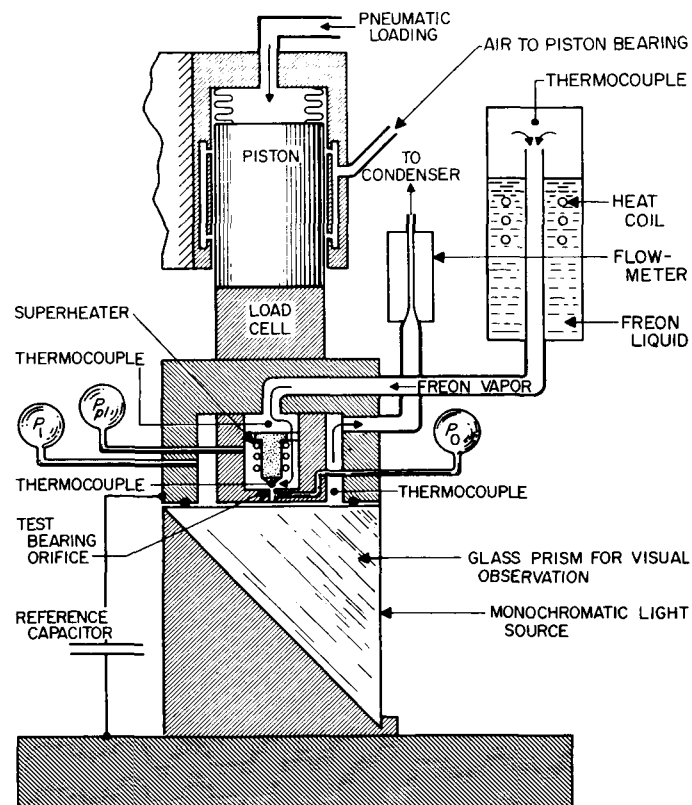


Fig. 7. Schematic diagram of closed-cycle-system test apparatus

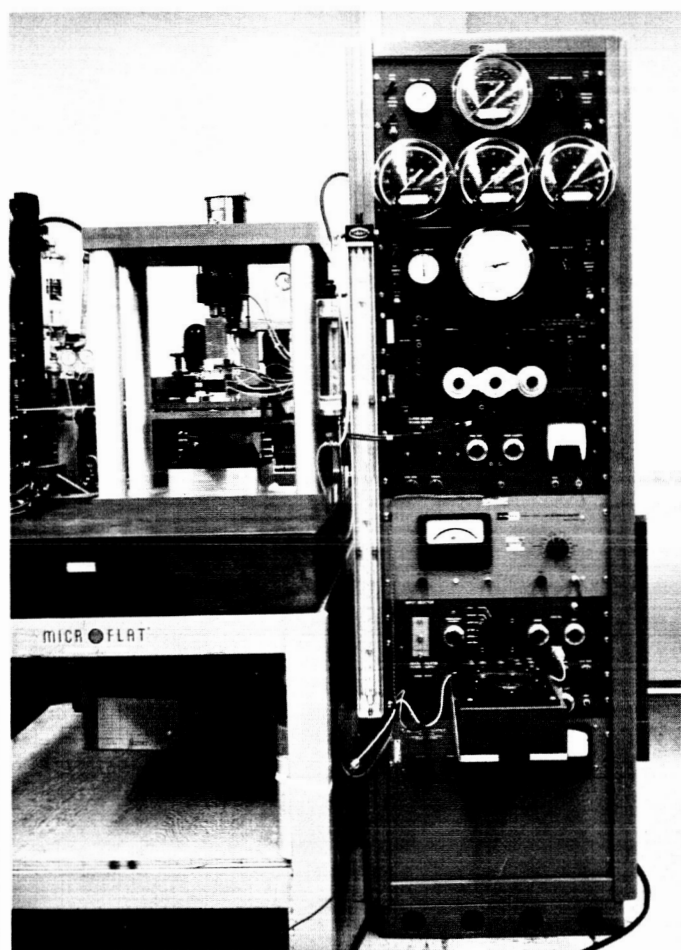


Fig. 8. Photograph of closed-cycle-system test apparatus

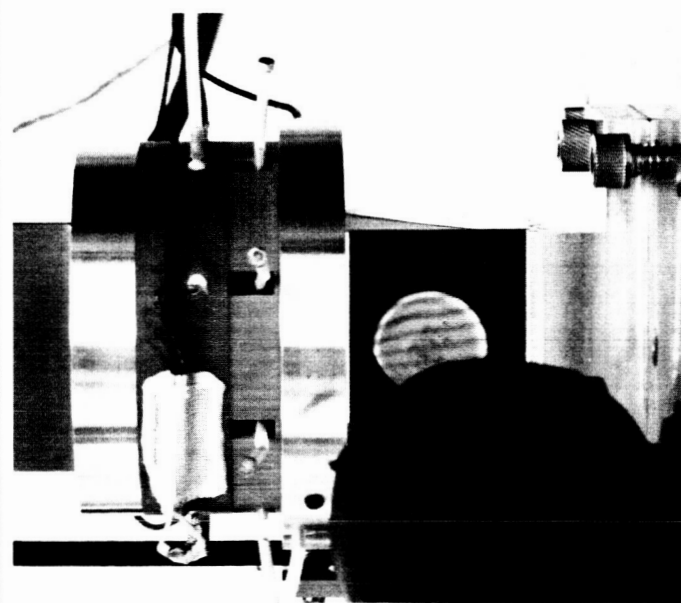


Fig. 9. View of interferometer showing a fringe pattern

cations. It consists of maintaining the evaporator temperature below the ambient and letting the saturated vapor become superheated as it flows through the tubing, at room temperature, to the bearing.

Freon C-318 was selected, because its pressure near room temperature is suitable for many gas-bearing applications. Its pressure is 10.5 psig at 45°F and increases approximately linearly to 30 psig at 76°F. The evaporator could be cooled by blowing off some of the Freon; it could be heated by applying electric current to the heater coil within the container. The temperature could thus be adjusted to the desired value. Several tests were made over this temperature range. The entrance to the pad bearing was observed visually as the evaporator temperature approached ambient in order to see when liquid flow occurred. In all instances in which the temperature of the evaporator was raised very slowly, liquid flow did not occur until the evaporator temperature was approximately 0.5°F above the bearing temperature. When the evaporator temperature was increased rapidly, a small amount of liquid flow would appear and then disappear, even at evaporator temperatures as much as 8°F below the bearing temperature. A possible explanation for this phenomenon is that small droplets were thrown upward into the open end of the evaporator discharge pipe (Fig. 7) as a result of vigorous boiling; the droplets were not vaporized as long as the temperature difference between the evaporator and the bearing was below 8°F, whereas, for greater temperature differences, complete vaporization occurred.

A number of tests were run to compare the actual mass flow rates of Freon C-318, as measured with the flowmeter, with two different calculated flows; i.e., the isentropic flow through the orifice and the isothermal viscous flow in the thin film of the bearing gap. Both flow calculations made use of a given gas constant  $R$ ; that is, the equation of state ( $p v = R T$ ) was assumed to apply. This assumption is not permissible over a large range of temperatures and pressures which include those near the saturation line (see Fig. 3). However, over a small range in the superheat region near the saturation line but considerably below the critical point, the equation of state gives values agreeing fairly well with those of thermodynamic charts for Freon.

The equations for the isentropic flow through the orifice are then:

$$W_{or} = \left( \frac{2}{k+1} \right)^{\frac{k-1}{2(k-1)}} \left( \frac{k g}{R T} \right)^{1/2} P_{pi} A C \quad (1a)$$

when

$$\frac{P_{pl}}{P_0} \leq \left( \frac{2}{k+1} \right)^{\frac{k}{k-1}}$$

$$W_{or} = \left[ \left( \frac{\left( \frac{P_{pl}}{P_0} \right)^{\frac{k-1}{k}} - 1}{\left( \frac{P_{pl}}{P_0} \right)^{\frac{k+1}{k}}} \right) \frac{2kg}{(k-1)RT} \right]^{1/2} P_{pl} AC \quad (1b)$$

when

$$\frac{P_{pl}}{P_0} > \left( \frac{2}{k+1} \right)^{\frac{k}{k-1}}$$

The equation for the isothermal viscous flow through the bearing gap is

$$W_B = \frac{\pi b^3 (P_0^2 - P_1^2)}{12 \mu RT \log \frac{R}{R_0}} \quad (2)$$

where

- $W_{or}$  = weight flow through control orifice
- $W_B$  = weight flow through pad bearing
- $k = C_p/C_v$  = ratio of specific heats
- $g$  = acceleration of gravity
- $T$  = absolute temperature
- $R$  = gas constant in units of length per degree absolute temperature
- $P_{pl}$  = absolute pressure upstream of control orifice
- $P_0$  = absolute pressure at bearing entrance
- $P_1$  = absolute pressure at bearing exit
- $A$  = orifice area
- $C$  = orifice coefficient
- $b$  = uniform gap between bearing faces
- $\mu$  = coefficient of absolute viscosity
- $R$  = bearing outer radius
- $R_0$  = bearing radius at inner edge of gap

The parameters of the bearing are:  $\log R/R_0 = 3.56$ ,  $A = 1.25 \times 10^{-5}$  in.<sup>2</sup>,  $C = 1$ ,  $g = 386$  in./sec<sup>2</sup>,  $P_1 = 14.7$  psia; and of Freon C-318:  $K = 1.05$ ;  $R = 93$  in.<sup>3</sup>/°R;  $\mu = 1.7 \times 10^{-9}$  lb-sec/in.<sup>2</sup>

The weight flow  $W_{or}$  has been calculated for two values of the supply pressure  $P_{pl}$  (viz. 38.7 and 30.7 psia) and is plotted in Fig. 10 (solid lines). The symbols in Fig. 10 represent measured points. The correlation is quite satisfactory.

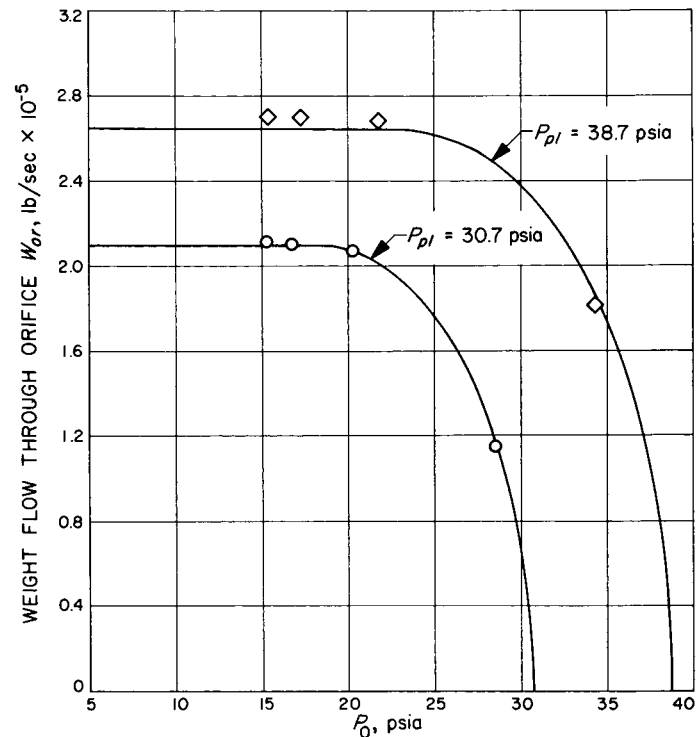


Fig. 10. Weight flow through orifice vs absolute pressure at bearing entrance

In order to check Eq. (2) for the flow  $W_B$  through the bearing gap, the two faces of the bearing were spaced apart by three small shims, approximately  $0.0625 \times 0.0625$  in., equally spaced near the outer edge of the bearing. The thickness of the shim stock was determined by micrometer measurements and computing the average of several thicknesses. The flow rates  $W_B$  were measured for four shim thicknesses (viz. 1.13, 0.73, 0.50, and 0.285 mil) and are represented by symbols in Fig. 11. A comparison with the solid lines, calculated from Eq. (2), shows that the performance of gas bearings, operating on Freon vapor within the limits investigated, can be predicted satisfactorily by the analytical treatment developed.

The Freon, after condensation, has to be pumped back into the evaporator to complete the cycle. This can be accomplished by a small mechanical pump whose power input is negligibly small. However, it is desirable to eliminate all moving or rotating elements from the system and to employ a technique such as capillary action which can be used in the zero-g environment of a spacecraft.

In order to study the feasibility of a capillary pump, the performance of wicks, consisting of bundles of parallel glass fibers, was investigated. Figure 12 shows how capillaries are formed by perfectly packed glass fibers.

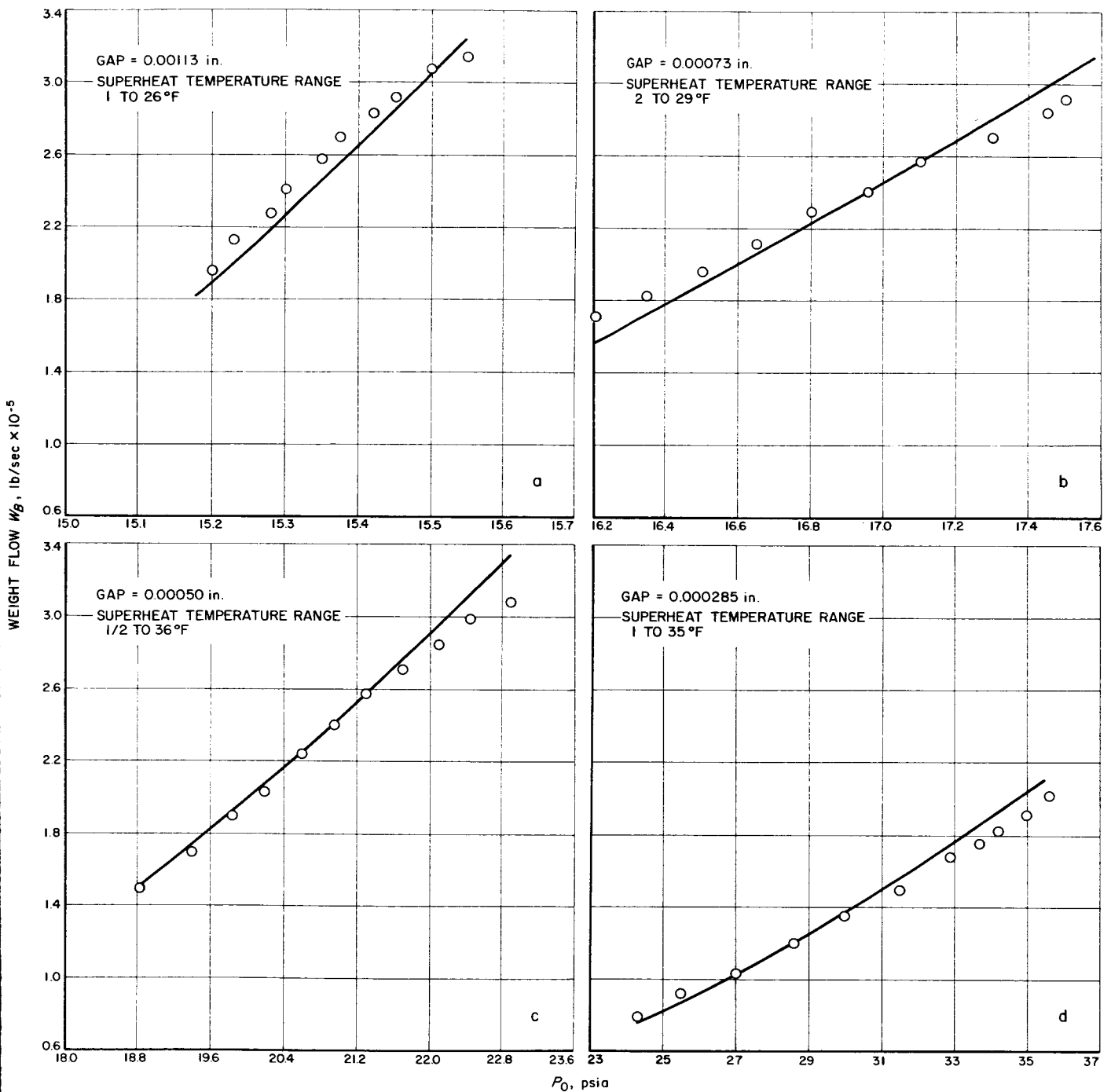


Fig. 11. Weight flow through bearing vs absolute pressure at bearing entrance

The rise  $Z$  of a liquid of weight density  $\gamma$  above the free surface in a capillary of circular cross-section with diameter  $d$  is given by

$$Z = \frac{4 \sigma \cos \theta}{\gamma d} \quad (3)$$

where  $\sigma$  is the surface tension and  $\theta$  the contact angle of the meniscus with the wall of the capillary. The pressure difference obtainable across the meniscus surface is

$$\Delta P = \frac{4 \sigma \cos \theta}{d} \quad (4)$$

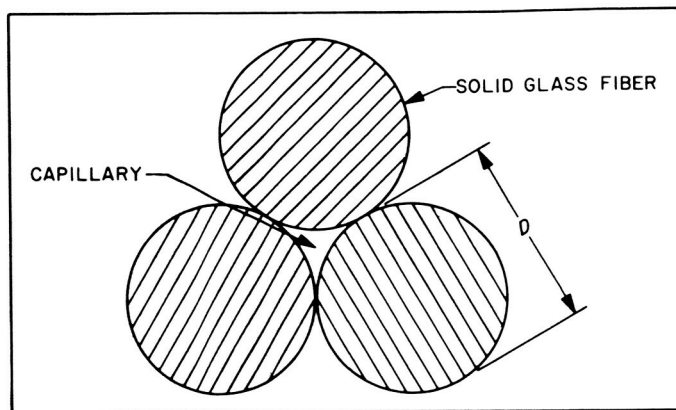
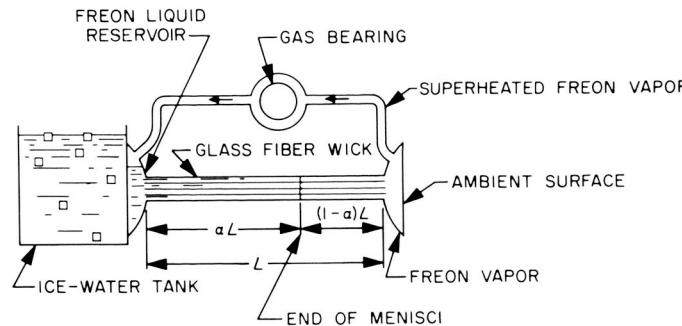


Fig. 12. Capillary formed by perfectly packed glass fibers

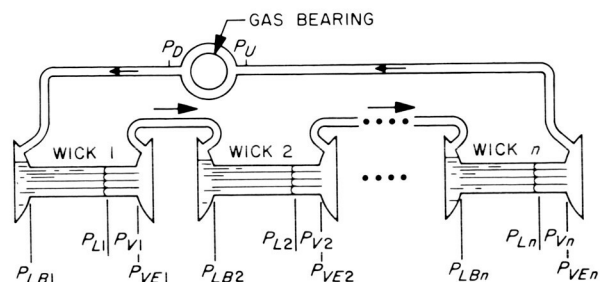
Thus, for a given liquid,  $\Delta P$  varies inversely with the capillary diameter  $d$ . It should be noted that  $\Delta P$  is not dependent on the weight density  $\gamma$ ; i.e., the capillary pressure is available under zero- $g$  or near-zero- $g$  conditions.

An analytical and experimental investigation of capillary pumping has been reported in detail by one of the authors (Refs. 2 and 3). The wicks studied consisted of 12-in.-long glass tubes of  $\frac{3}{8}$ -in. ID, compactly filled with 0.0005-in.-D glass fibers. The fibers flare out at each end and partially fill bell-shaped cavities (see Fig. 12). One end is connected to a wall of a container of ice water, the other end to a glass plate containing a heating element. The apparatus was charged with Freon-114 vapor.

The observed pressure  $\Delta P$  was in all cases substantially lower than the value predicted on the assumption of a perfectly packed wick, yet high enough for the operation of gas bearings in low- $g$  environments. The difference can probably be explained by the imperfect packing of the fibers. The pressure can be raised, where necessary, by a multiple-stage arrangement, as indicated in Fig. 13. A hemispherical gas bearing with a rotor weighing 64 g was successfully operated on both a single-stage and double-stage capillary pump in a closed system using Freon-114 vapor (Fig. 14). Further tests with improved wicks are planned; the results will be published later.



a. SINGLE-STAGE CAPILLARY PUMP



b. MULTIPLE-STAGE CAPILLARY PUMP

Fig. 13. Single-stage and multiple-stage capillary pumps

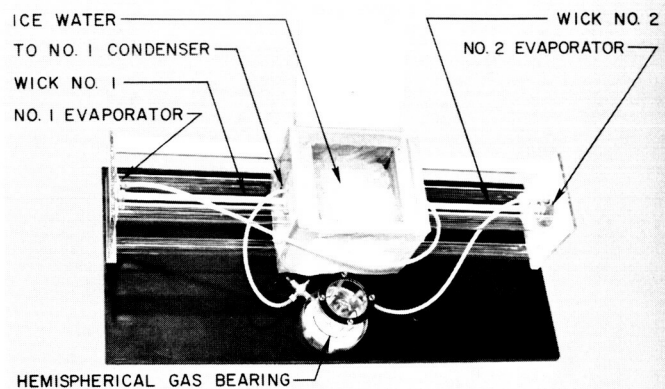


Fig. 14. Experimental closed-cycle system for hemispherical bearing

## II. CONCLUSION

It has been demonstrated that closed-cycle gas-supply systems, using Freon or a two-phase fluid of similar thermodynamic characteristics, are feasible for the operation of gas bearings and other devices. The pump-

ing can be accomplished by thermal input and capillary action without rotating mechanical components. Operation in zero-*g* or near-zero-*g* environments is possible.

## ACKNOWLEDGMENT

The authors wish to express their appreciation to Mr. William T. Jones for his contributions to this project.

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